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QUASI EQUILIBRIUM FUEL-AIR HEAT BALANCED CYCLE ANALYSIS.(U)
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QUASI EQUILIBRIUM FUEL-AIR HEAT BALANCED CYCLE ANALYSIS

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Annapolis, Maryland

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The quasi equilibrium thermodynamic model of the Naval Academy Heat Balanced Engine (NAHBE) has been modified to include the influence of fuel-air chemistry on predicted indicated engine performance. Heat addition to the Air Standard Heat Balanced Cycle was expressed in terms of an appropriate fuel-air ratio and heating value for a standard fuel. Indicated parameters including mean effective pressure, peak pressure, specific fuel consumption and thermal efficiency for compatible Otto and Heat Balanced cycles were calculated and

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compared. Performance parameters for both cycles were obtained at equal compression ratios, fuel-air ratios, fuel type, and engine rpm. Results show that for overall stoichiometric heat addition the Heat Balanced cycle can produce greater indicated engine power, higher indicated thermal efficiency and lower indicated specific fuel consumption than the corresponding Otto cycle. Further analysis indicates that the optimum heat balancing conditions occur for constant volume heat addition with rich mixture composition followed by constant pressure heat addition with lean composition.

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I. ABSTRACT

The quasi equilibrium thermodynamic model of the Naval Academy Heat Balanced Engine (NAHBE) has been modified to include the influence of fuel-air chemistry on predicted indicated engine performance. Heat addition to the Air Standard Heat Balanced Cycle was expressed in terms of an appropriate fuel-air ratio and heating value for a standard fuel. Indicated parameters including mean effective pressure, peak pressure, specific fuel consumption and thermal efficiency for compatible Otto and Heat Balanced cycles were calculated and compared. Performance parameters for both cycles were obtained at equal compression ratios, fuel-air ratios, fuel type, and engine rpm. Results show that for overall stoichiometric heat addition the Heat Balanced cycle can produce greater indicated engine power, higher indicated thermal efficiency and lower indicated specific fuel consumption than the corresponding Otto cycle. Further analysis indicates that the optimum heat balancing conditions occur for constant volume heat addition with rich mixture composition followed by constant pressure heat addition with lean composition.

II. INTRODUCTION

The experimental performance of a variety of actual heat balanced internal combustion engines has been shown to depend significantly on the fuel-air characteristics of the particular applications.[1,2,3] Any I.C. engine performance depends upon, among other parameters, the air-fuel ratio; fuel type, i.e., gasoline, alcohol, diesel fuel oil; fuel heating value; and octane or cetane index. The thermodynamic basis for predicting the influence of these factors on standard cycles such as the Otto, Diesel or Dual cycles, is covered in a variety of I.C. engine texts.[4,5,6,7] Spark and/or compression ignition engine behavior can be predicted using an air standard cycle or a fuel-air cycle analysis. In the air standard model the charge in the engine is treated as being air alone and heat addition and rejection are used to represent the combustion and exhaust processes. In the fuel-air analysis, the thermochemistry of the engine fuel-air compression and combustion processes, as well as the expansion and exhaust processes of the combustion products are treated in detail. This method allows a particular engine to be considered analytically in such a way as to produce more realistic indicated engine performance predictions than results based on air standard cycle calculations. In this study the preliminary rationale for a fuel-air heat balanced cycle analysis will be developed.

III. HEAT BALANCED CYCLE: PRELIMINARY FUEL-AIR ANALYSIS

A thermodynamic model for the unique heat balanced engine concept has been reported in earlier studies.[8] Figure 1 and Table 1 summarize the particular features relevant to the theoretical heat balanced cycle analysis. In this original heat engine it is assumed that the total mass can be separated into two nonmixing regions; the total heat addition consists of separate constant volume and constant pressure processes, and all thermodynamic processes are quasi-equilibrium in nature. Based on calculations for this air standard cycle it was shown that the quasi-equilibrium heat balanced cycle (QEHBC) for optimum configuration can achieve a greater thermal efficiency and lower peak pressure than the comparable Otto cycle.

These results were obtained from a parametric study of the governing equations for the ideal air standard Otto, Diesel, Dual and Heat Balanced cycles. All cycles were compared on the basis of congruent geometries, equal total masses, and identical total heat additions. At an 8:1 compression ratio two additional heat balanced cycle configurations were investigated. In the first study the mass in the two regions was unchanged while the percentage of the total heat added to these separate masses was varied. In the second treatment the heat addition to the two regions was held constant and the fraction of the total mass in the two regions was changed. Based on these calculations it was shown that for a given compression ratio, total mass, and total heat addition QEHBC performance can be changed dramatically by simply varying the heat and mass balancing ratios.

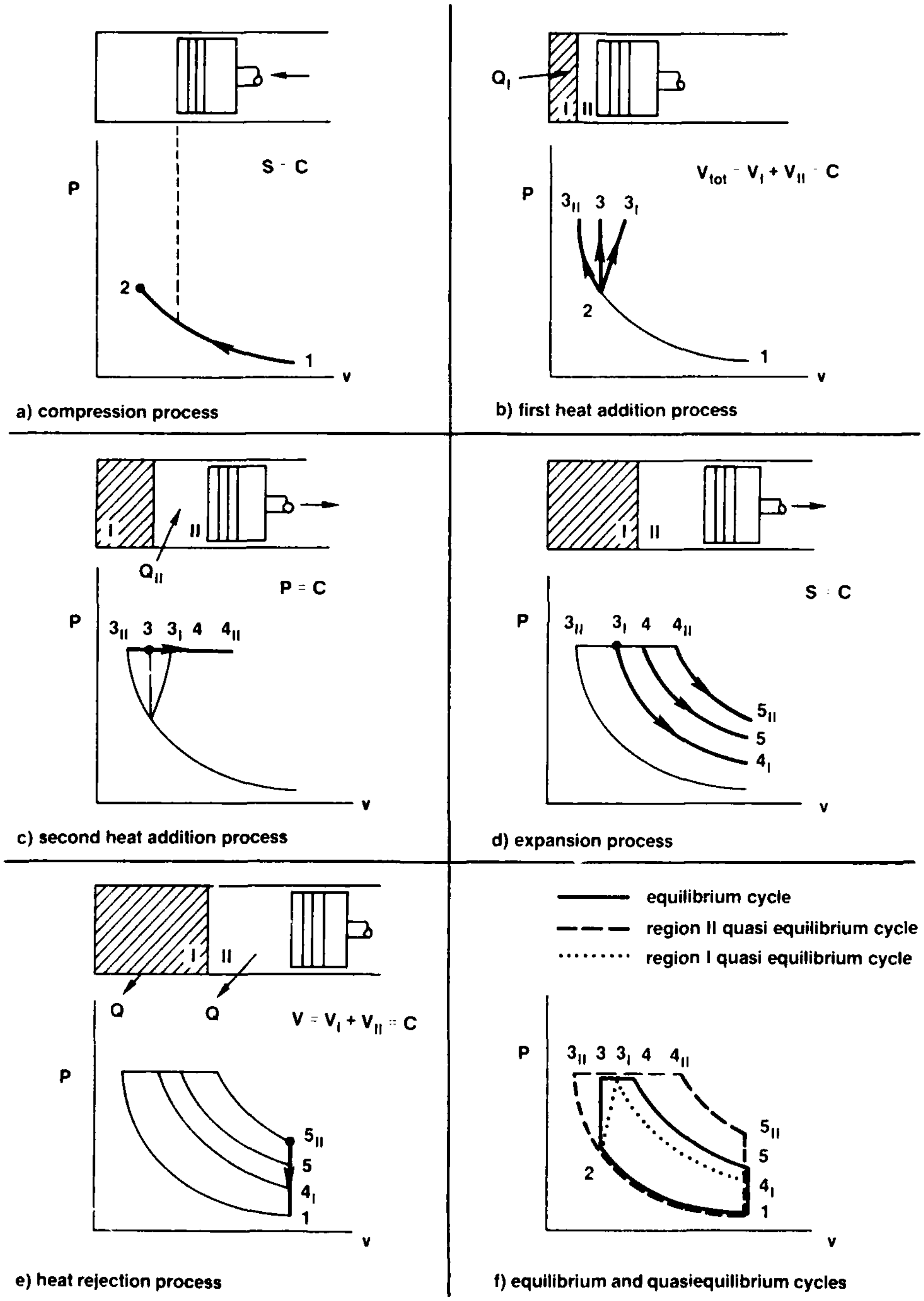

Figure 1 QEHBC THERMODYNAMIC P-V DIAGRAM

TABLE I. QEHBC THERMODYNAMIC CYCLE*

State: Initial		State: Final	Process: Region I	Process: Region II
			compression process	
1_{I}	→	2_{I}	isentropic	-------
1_{II}	→	2_{II}	--------	isentropic
			first heat addition	
2_{I}	→	3_{I}	constant TOTAL volume heat addition	--------
2_{II}	→	3_{II}	---------	isentropic compression
			second heat addition	
3_{II}	→	4_{II}	---------	constant pressure heat addition
			expansion process	
3_{I}	→	4_{I}	isentropic	----------
4_{II}	→	5_{II}	---------	isentropic
			heat rejection	
4_{I}	→	1_{I}	constant volume	----------
5_{II}	→	1_{II}	--------	constant volume

*To attain the QEHBC cycle, regions I and II must remain separated throughout the cycle and all processes are quasi-equilibrium in nature. If the two regions mix and attain equilibrium the cycle produces the classical combined or dual cycle.

For any heat balanced cycle the total mass and heat added to the thermodynamic cycle can be expressed in terms of the two regions as

$$M_t = M_I + M_{II} \tag{1}$$

where

M_t = total mass, lbm

M_I = mass in region I, lbm

M_{II} = mass in region II, lbm

and

$$Q_t = Q_I + Q_{II} \tag{2}$$

with

Q_t = total heat addition, Btu

Q_I = heat addition to region I, Btu

Q_{II} = heat addition to region II, Btu

Two balancing ratios, the mass balancing ratio, β_M, and the heat balancing ratio, β_H, have been previously defined[8] for the heat balanced cycle and are given by the relations

$$\beta_M = \frac{M_I}{M_{II}} \tag{3}$$

and

$$\beta_H = \frac{Q_{II}}{Q_I} \tag{4}$$

Using Equations (3) and (4) the mass in regions I and II can be expressed in terms of β_M and the total mass, M_t, as

$$M_I = \left[\frac{\beta_M}{1 + \beta_M}\right] M_t \qquad (5)$$

and

$$M_{II} = \left[\frac{1}{1 + \beta_M}\right] M_t \qquad (6)$$

Air standard cycle analysis can be modified to approximate the behavior of an actual engine by expressing the thermodynamic cycle heat addition in terms of the actual engine fuel-air requirements. Thus the total heat addition is calculated in terms of the fuel heating value, engine fuel-air ratio, and the fuel-air charge induced into the engine. The total mass in the engine ideally equals the conditions at the end of the intake stroke at BDC, equal to

$$M_t = \frac{P_1 V_1}{RT_1} \qquad (7)$$

where

P_1 = inlet pressure, lbf/ft^2

V = volume @ BDC, ft^3

R_1 = gas constant for air

= 53.34 ft lbf/lbm R

T_1 = inlet temperature, ^oR

The volume at BDC consists of the clearance volume, V_c, and the displacement volume, V_D, with the total volume given as

$$V_1 = V_c + V_D \qquad (8)$$

The compression ratio, r_v, a ratio of the volume of BDC to the volume at TDC, can be written as

$$r_v = \frac{V_{BDC}}{V_{TDC}} = \frac{V_c + V_D}{V_c} \qquad (9)$$

The total volume in terms of the compression ratio and displacement volume is then

$$V_1 = V_D\left[\frac{r_v}{r_v - 1}\right] \tag{10}$$

The total volume at BDC of an actual I.C. engine cannot be completely filled with fuel and air, i.e., the induced charge. This is due to the presence of residual gases remaining from the exhaust stroke. Taylor[4] suggests that the residuals and charge at BDC have approximately the same density and the displacement volume can be considered to be filled with reactants and the clearance volume to be filled with residual gases. The total mass of air in the air standard cycle that should be used to determine the required heat addition is thus expressed as

$$M_c = M_t\left[\frac{V_D}{V_1}\right] = M_t\left[\frac{r_v - 1}{r_v}\right] \tag{11}$$

where

M_c = mass of air associated with combustion, lbm

M_t = total mass of air, lbm

r_v = compression ratio

The appropriate heat addition to an air standard cycle in terms of the engine fuel-air characteristics is given as

$$Q_t = \left[\frac{HV \times M_c}{AF_t}\right] \tag{12}$$

where

Q_t = total heat addition, Btu

HV = fuel heating value, Btu/lbm fuel

AF_t = air-fuel ratio, lbm air/lbm fuel

M_c = mass of air, lbm

Substituting Equation (11) into Equation (12) yields the proper expression for the total heat addition for the air standard cycle in terms of engine air-fuel requirements as

$$Q_t = \left[\frac{HV \times M_t}{AF_t}\right]\left[1 - \frac{1}{r_v}\right] \tag{13}$$

Since the total heat addition in the heat balanced cycle consists of portions added separately to regions I and II expressions for these two heat additions can also be written as

$$Q_I = \left[\frac{HV \times M_I}{AF_I}\right]\left[1 - \frac{1}{r_v}\right] \tag{14}$$

and

$$Q_{II} = \left[\frac{HV \times M_{II}}{AF_{II}}\right]\left[1 - \frac{1}{r_v}\right] \tag{15}$$

Substituting Equations (13), (14), and (15) into Equation (2), relating Q_t to Q_I and Q_{II} yields

$$\left[\frac{HV \times M_t}{AF_t}\right] = \left[\frac{HV \times M_I}{AF_I}\right] + \left[\frac{HV \times M_{II}}{AF_{II}}\right] \tag{16}$$

Since the same fuel is associated with regions I and II, Equation (16) reduces tc

$$\frac{M_t}{AF_t} = \frac{M_I}{AF_I} + \frac{M_{II}}{AF_{II}} \tag{17}$$

Substituting Equations (5) and (6) for M_I and M_{II}, respectively gives

$$\frac{M_t}{AF_t} = \left[\frac{\beta_M}{1 + \beta_M}\right] \frac{M_t}{AF_I} + \left[\frac{1}{1 + \beta_M}\right] \frac{M_t}{AF_{II}}$$

$$(1 + \beta_M) = \beta_M \left[\frac{AF_t}{AF_I}\right] + 1 \left[\frac{AF_t}{AF_{II}}\right]$$

$$\beta_M \left[1 - \frac{AF_t}{AF_I}\right] = \left[\frac{AF_t}{AF_{II}} - 1\right]$$

or

$$\beta_M = \left[\frac{(AF_t/AF_{II}) - 1}{1 - (AF_t/AF_I)}\right] \tag{18}$$

Equation (18) indicates that the mass balancing ratio, β_M, for the heat balanced cycle should be a function of the total overall air-fuel ratio, AF_t, and the air-fuel ratios in regions I and II, AF_I and AF_{II}, respectively.

The heat balancing ratio, β_M, given by Equation (4) can now be rewritten using Equations (14) and (15) giving

$$\beta_H = \frac{Q_{II}}{Q_I} = \left[\frac{HV \times M_{II}}{AF_{II}}\right]\left[\frac{AF_I}{HV \times M_I}\right]$$

$$\beta_H = \frac{1}{\beta_M} \left[\frac{AF_I}{AF_{II}}\right] \tag{19}$$

Substituting Equation (18) for β_M in Equation (19) gives the result

$$\beta_H = \left[\frac{AF_I}{AF_{II}}\right] \left[\frac{1 - (AF_t/AF_I)}{(AF_t/AF_{II}) - 1}\right]$$

$$\beta_H = \frac{AF_I - AF_t}{AF_t - AF_{II}} = \frac{(AF_I/AF_t) - 1}{1 - (AF_{II}/AF_t)} \tag{20}$$

IV. PARAMETRIC ANALYSIS: HEAT BALANCED CYCLE HEAT ADDITION

Equation (20) shows that the heat balancing ratio, β_H, for the heat balanced cycle should also be a function of the overall air fuel ratio, AF_t, and the air-fuel ratios in the two regions I and II, AF_I and AF_{II}.

An inspection of Equations (18) and (20) reveals how the heat balanced cycle might produce the wide variation in performance suggested by the previous cycle studies even for the case of a specified compression ratio and fixed total mass and heat addition. Equation (18) shows that for a particular overall stoichiometry, AF_t, the air-fuel ratios in regions I and II, AF_I and AF_{II}, can be varied in such a way that the mass balancing ratio, β_M, will remain constant. These changes, however, would result in a variation in the heat balancing ratio, β_H. Similarly, Equation (20) suggests that the air-fuel variations can be made such that the heat balancing ratio, β_H, will remain constant with changes found in the corresponding mass balancing ratio, β_M. Equations (18) and (20) further indicate that the two balancing ratios, β_M and β_H, are not totally independent but relate to each other through the air fuel ratios in the separated regions I and II.

Tables II-VI give the results for a four stroke air standard heat balanced cycle. The following conditions were assumed in all calculations:

$$\text{fuel} = C_8H_{18} \text{ (octane)}$$

$$D \times L = 4" \times 4"$$

$$\text{inlet } P_1 = 14.7 \text{ psia}$$

$$\text{inlet } T_1 = 70^oF$$

$$AF_t = \text{stoichiometric} = 15.12$$

$$HV = 20604 \text{ Btu/lbm fuel}$$

$$N = 1300 \text{ rev/min}$$

$$\# \text{ cylinders} = 1$$

$$r_v = 8:1$$

$$\beta_M = 0.2 \rightarrow 2.0$$

$$\beta_H = 0.2 \rightarrow 2.0$$

The stoichiometric reaction of C_8H_{18} and air is equal to

$$C_8H_{18} + 12.5[O_2 + 3.76\ N_2] \rightarrow 8CO_2 + 9H_2O + 47\ N_2$$

where the total air-fuel ratio is given by

$$AF_t = \left[\frac{(12.5)(4.76)(28.97)\ \text{lbm air}}{(1.0)(114)\ \text{lbm fuel}}\right] = 15.12\ \frac{\text{lbm air}}{\text{lbm fuel}}$$

The mass flow rate of air associated with the required engine heat addition calculation is given by the expression

$$\dot{M}_c = \frac{M_c \times N}{2} = \frac{M_t N}{2}\left[1 - \frac{1}{r_v}\right] \tag{21}$$

where

$\dot{M}_c$ = combustion air mass flow rate lbm air/min

M_c = mass of combustion air, lbm air/intake stroke

M_t = total mass of air, lbm

N = engine revolution, rev/min

r_v = compression ratio

2 = revolutions/intake stroke

The mass flow rate of fuel required for the proper heat addition is then equal to

$$\dot{M}_f = \frac{M_t N}{2AF_t}\left[1 - \frac{1}{r_v}\right] \tag{22}$$

where

$\dot{M}_f$ = fuel mass flow rate, lbm fuel/min

AF_t = total overall air-fuel ratio, lbm air/lbm fuel.

The indicated mean effective pressure is equal to the net cycle work divided by the displacement volume, or

$$\overline{P} = \frac{W_{net}}{V_D} \tag{23}$$

with

$\overline{P}$ = indicated mean effective pressure lbf/ft²

W_{net} = net cycle work, ft lbf

V_D = displacement volume, ft³

The indicated power for the cycle can be obtained using the mean effective pressure then as the standard relationship

$$\dot{W} = \frac{\overline{P}\ L\ A\ N\ c}{33{,}000\ n} \tag{24}$$

where

$\dot{W}$ = indicated engine power, hp

$\overline{P}$ = mean effective pressure, lbf/ft²

L = stroke, ft/stroke

A = piston cross-sectional area, ft²

N = engine speed, rev/min

c = number of cylinders = 1

33,000 = ft lbf/hp-min

n = number of revolutions/stroke = 2

The indicated thermal efficiency for the cycle can be calculated from the expression

$$\eta = \left[\frac{\dot{W}}{\dot{M}_f HV} \right] \tag{25}$$

where

$\dot{W}$ = indicated engine power, Btu/min

$\dot{M}_f$ = fuel mass flow rate, lbm fuel/min

HV = fuel heating value, Btu/lbm fuel

Using Equations (22) and (24) an expression for the indicated specific fuel consumptions can be written as

$$ISFC = \frac{\dot{M}_f}{\dot{W}} \tag{26}$$

with

ISFC = indicated specific fuel consumption, lbm/hp-hr

$\dot{M}_f$ = fuel mass flow rate, lbm fuel/hr

$\dot{W}$ = indicated engine power, hp

Numerical results obtained using Equations (21)-(26) are found in Tables II-VI and are presented graphically in Figures 2-6. Table II and Figure 2 show the predicted indicated mean effective pressure for various heat and mass balancing ratios. Table II shows that for the particular values of β_M and β_H the QEHBC in all cases exceeded the output of 311.38 psi for the corresponding Otto cycle. Figure 2 also indicates that ideal power output reaches its maximum values for heat and mass balancing ratios less than one.

TABLE II. INDICATED MEAN EFFECTIVE PRESSURE vs. BALANCING RATIOS

(311.38 psi)*

β_H \ β_M	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.2	371.00	368.06	365.52	363.359	361.50	359.90	358.5	357.28	356.19	355.23
0.4	374.97	373.25	371.65	370.23	368.98	367.88	366.90	366.04	365.28	364.59
0.6	375.14	374.35	373.46	372.60	371.80	371.07	370.41	369.81	369.28	368.79
0.8	373.31	373.27	372.94	372.52	372.07	371.63	371.22	370.83	370.47	370.14
1.0	370.40	370.97	371.11	371.05	370.89	370.69	370.47	370.25	370.04	369.84
1.2	366.92	368.00	368.53	368.77	368.86	368.86	368.81	368.74	368.64	368.55
1.4	363.14	364.68	365.55	366.05	366.34	366.51	366.61	366.65	366.66	366.66
1.6	359.26	361.19	362.35	363.07	363.55	363.87	364.09	364.24	364.34	364.41
1.8	355.36	357.64	359.06	359.99	360.62	361.07	361.40	361.64	361.82	361.96
2.0	351.52	345.11	355.76	356.86	357.64	358.20	358.63	358.95	359.20	359.40

*Results obtained for equivalent Otto cycle.
(maximum value obtained)

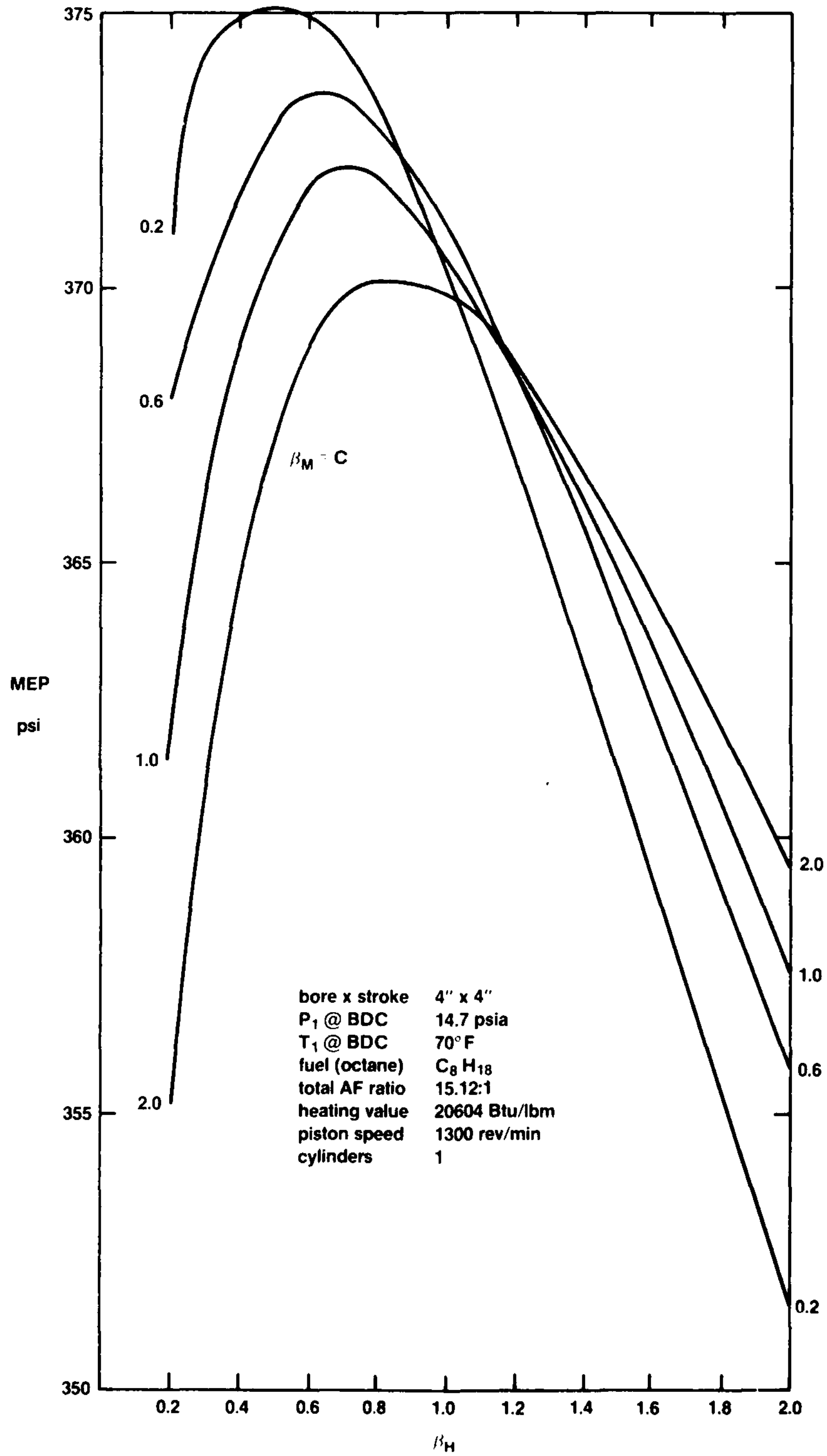

Figure 2 INDICATED MEP VS BALANCING RATIOS

TABLE III. INDICATED THERMAL EFFICIENCY vs. BALANCING RATIOS

(56.484%)*

β_H \ β_M	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.2	67.300	66.765	66.305	65.913	65.576	65.284	65.031	64.809	64.613	64.439
0.4	68.019	67.706	67.417	67.159	66.932	66.732	66.556	66.400	66.261	66.136
0.6	68.050	67.906	67.745	67.588	67.443	67.310	67.191	67.083	66.986	66.898
0.8	67.718	67.709	67.651	67.574	67.492	67.413	67.338	67.268	67.203	67.143
1.0	67.190	67.293	67.318	67.307	67.278	67.242	67.203	67.163	67.125	67.088
1.2	66.558	66.755	66.851	66.895	66.910	66.910	66.901	66.888	66.871	66.854
1.4	65.873	66.152	66.309	66.400	66.454	66.485	66.502	66.510	66.512	66.511
1.6	65.168	65.519	65.729	65.861	65.947	66.005	66.044	66.072	66.090	66.103
1.8	64.462	64.875	65.132	65.300	65.416	65.497	65.556	65.600	65.633	65.658
2.0	63.765	64.235	64.533	64.734	64.875	64.977	65.054	65.112	65.158	65.194

*Results obtained for equivalent Otto cycle.
(maximum value obtained)

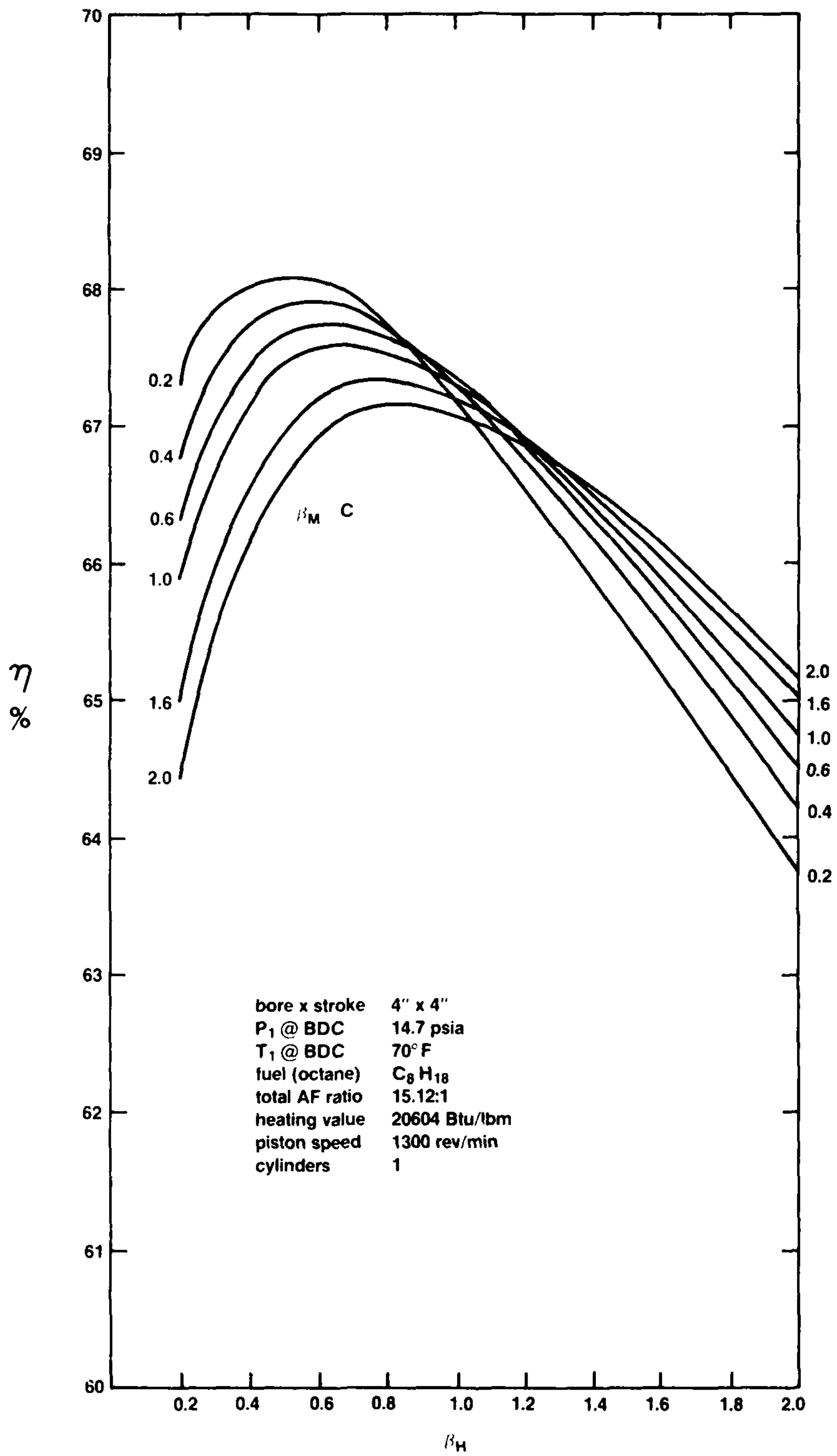

Figure 3 THERMAL EFFICIENCY VS BALANCING RATIOS

TABLE IV. ISFC vs. BALANCING RATIO

(0.21868 lbm fuel/hp-hr)*

β_H \ β_M	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.2	0.18354	0.18501	0.18629	0.18739	0.18836	0.18920	0.18994	0.19059	0.19117	0.19169
0.4	0.18160	0.18244	0.18322	0.18392	0.18455	0.18510	0.18559	0.18602	0.18641	0.18676
0.6	0.18151	0.18190	0.18233	0.18275	0.18315	0.18351	0.18383	0.18413	0.18440	0.18464
0.8	0.18240	0.18243	0.18258	0.18279	0.18301	0.18323	0.18343	0.18362	0.18380	0.18396
1.0	0.18364	0.18355	0.183486	0.18352	0.18360	0.18369	0.18380	0.18391	0.18401	0.18411
1.2	0.18558	0.185033	0.18477	0.18465	0.18460	0.18460	0.18463	0.18467	0.18471	0.18476
1.4	0.18751	0.18672	0.18628	0.18602	0.18587	0.18579	0.18574	0.18572	0.18571	0.18571
1.6	0.18954	0.18853	0.18792	0.18755	0.1873	0.18714	0.18702	0.18695	0.18689	0.18686
1.8	0.19162	0.19040	0.18964	0.18916	0.18882	0.18859	0.18842	0.18829	0.18820	0.18812
2.0	0.19371	0.19229	0.19141	0.19081	0.19046	0.19010	0.18987	0.18970	0.18957	0.18946

*Results obtained for equivalent Otto Cycle.
(maximum value obtained)

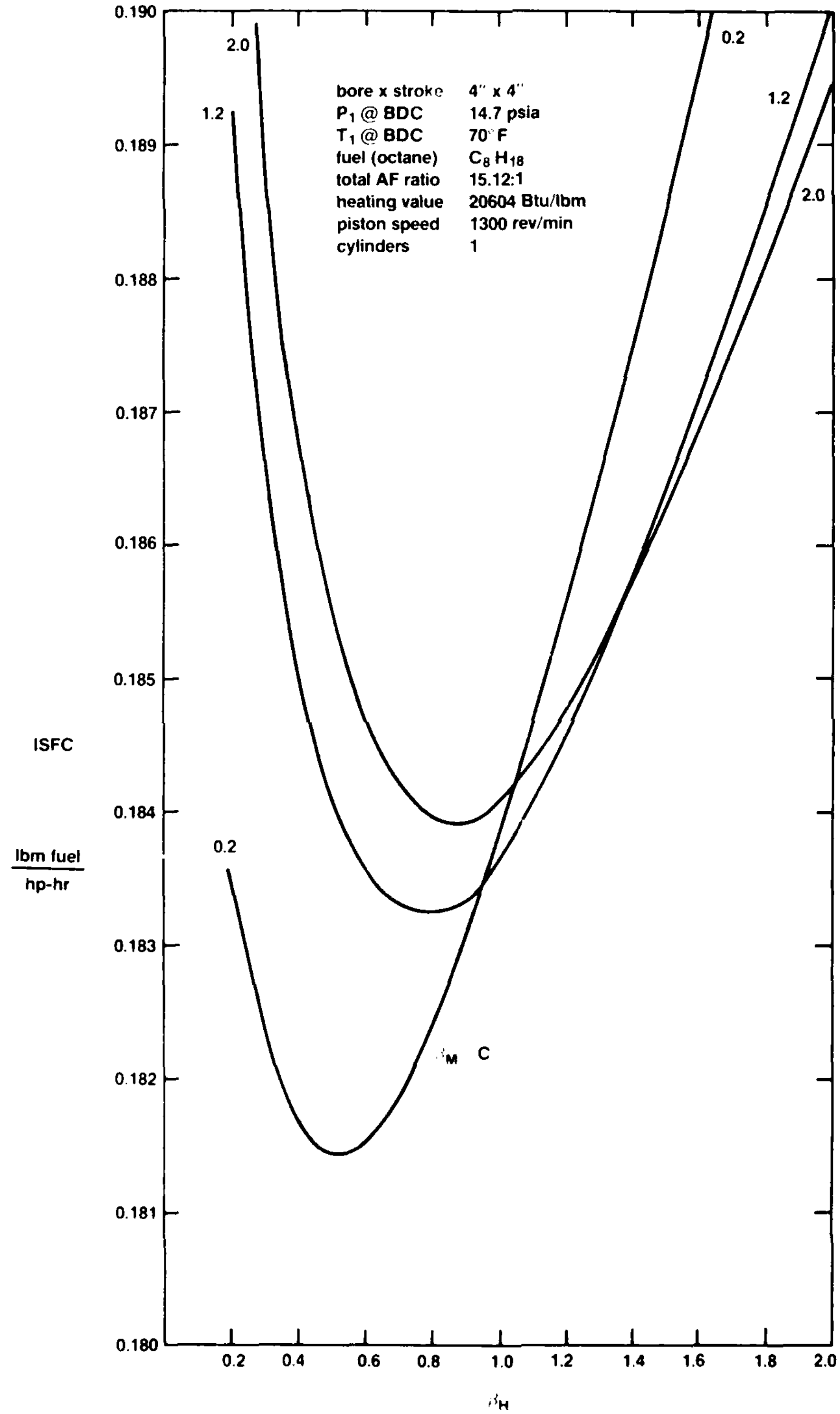

Figure 4 ISFC VS BALANCING RATIOS

Indicated thermal efficiency calculations for the heat balanced cycle are listed in Table III and plotted in Figure 3. Table III and Figure 3 reveal that the thermal efficiency is greater than Otto cycle value of 57.484% for all values of β_M and β_H. A comparison of Figures 2 and 3 further shows that the mean effective pressure as well as thermal efficiency is larger than those for the Otto cycle for heat balanced cycles having β_M and β_H less than one.

The indicated specific fuel consumption predictions are presented in Table IV and graphically shown in Figure 4. Table IV suggests that the Otto cycle specific fuel consumption of 0.21868 lbm fuel/hp-hr is greater than all heat balanced cycle values. Figure 4 shows that the minimum fuel consumption rates occur at values of β_M and β_H less than one.

These results of improved performance are in agreement with heat balanced engine predictions reported in earlier studies.[1,2,8] The required air fuel ratios for the two separate heat additions are found in Tables V and VI and plotted in Figures 5 and 6. A study of Table V and Figure 5 reveals that the air fuel ratio for region I, i.e., composition associated with constant total volume heat addition, is rich for all values of β_M and β_H less than one and lean for all values of β_M and β_H greater than one. Likewise Table VI and Figure 6 suggest that the air fuel ratio for region II, mass associated with constant pressure heat addition, is lean for all values of β_M and β_H less than one and rich for all values of β_M and β_H greater than one. In addition, the conditions of

TABLE V. $AF)_I$ vs. BALANCING RATIO

(15.12 lbm air/lbm fuel)*

β_H \ β_M	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.2	3.024	5.184	6.804	8.064	9.072	9.897	10.584	11.166	11.664	12.096
0.4	3.528	6.048	7.938	9.408	10.584	11.546	12.348	13.027	13.608	14.112
0.6	4.032	6.912	9.072	10.752	12.096	13.196	14.112	14.887	15.552	16.128
0.8	4.536	7.776	10.206	12.096	13.608	14.845	15.876	16.748	17.496	18.144
1.0	5.04	8.64	11.34	13.44	15.12	16.495	17.64	18.609	19.44	20.16
1.2	5.544	9.504	12.474	14.78	16.632	18.144	19.404	20.470	21.384	22.176
1.4	6.048	10.368	13.608	16.128	18.144	19.794	21.168	22.331	23.328	24.192
1.6	6.552	11.232	14.742	17.472	19.656	21.443	22.932	24.192	25.272	26.208
1.8	7.056	12.096	15.876	18.816	21.168	23.092	24.696	26.053	27.22	28.22
2.0	7.56	12.96	17.01	20.16	22.68	24.742	26.46	27.914	29.16	30.24

*Results obtained for equivalent Otto Cycle.
(total stoichiometric charge)

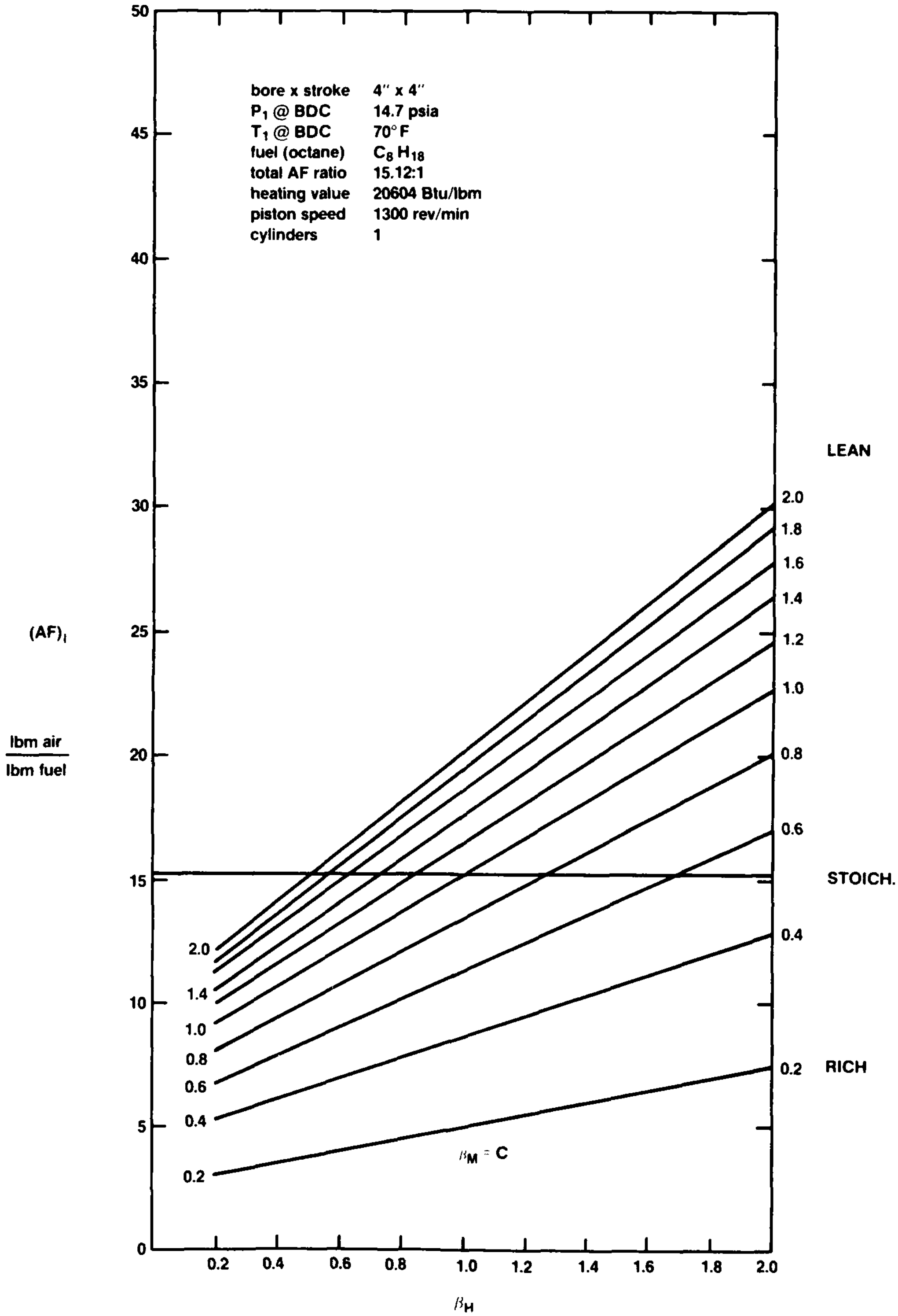

Figure 5 REGION I AF RATIO VS BALANCING RATIOS

TABLE VI. $AF)_{II}$ vs. BALANCING RATIO

(15.12 lbm air/lbm fuel)*

β_H \ β_M	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.2	75.6	64.8	56.7	50.4	45.36	41.236	37.8	34.892	32.4	30.24
0.4	44.1	37.8	33.075	29.4	26.46	24.055	22.05	20.354	18.9	17.64
0.6	33.6	28.8	25.20	22.4	20.16	18.327	16.8	15.508	14.4	13.44
0.8	28.35	24.3	21.263	18.9	17.01	15.464	14.175	13.085	12.15	11.34
1.0	25.2	21.6	18.90	16.8	15.12	13.746	12.6	11.631	10.8	10.08
1.2	23.1	19.8	17.325	15.4	13.86	12.6	11.55	10.662	9.9	9.24
1.4	21.6	18.514	16.2	14.4	12.96	11.782	10.8	9.969	9.257	8.64
1.6	20.475	17.55	15.356	13.65	12.285	11.168	10.238	9.45	8.775	8.19
1.8	19.6	16.8	14.7	13.067	11.76	10.691	9.8	9.046	8.4	7.84
2.0	18.9	16.2	14.175	12.6	11.34	10.309	9.45	8.723	8.1	7.56

*Results obtained for equivalent Otto cycle.
(total stoichiometric charge)

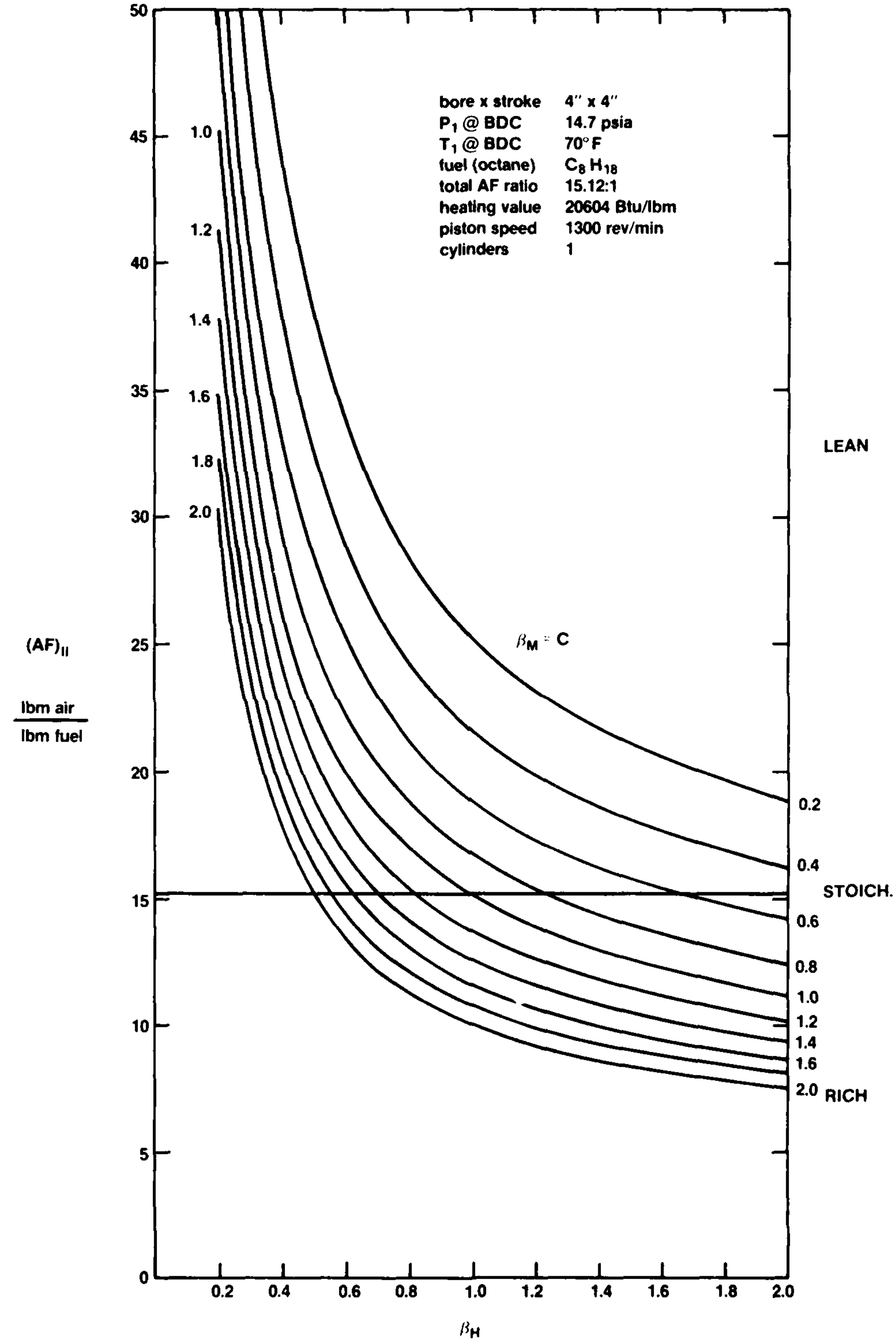

Figure 6 REGION II AF RATIO VS BALANCING RATIOS

optimum thermodynamic performance, i.e., small balancing ratios cited previously, correspond to a rich constant total volume heat addition follows by a lean constant total pressure heat addition.

The use of an Air Standard cycle to predict the performance of the actual engine precludes the effect of variable specific heats of the charge, residual gases in the clearance volume, and the thermochemistry of the actual fuel-air combustion process. In this sense the previous calculations merely suggest the trends anticipated in the proper design of a heat balanced engine. To further assess the influence of the actual fuel-air properties on β_M and β_H the total charge of reactants in the displacement volume next were treated as an ideal fuel-air gaseous mixture where

$$N_{0_t} = N_{0_I} + N_{0_{II}} \tag{27}$$

$$N_{f_t} = N_{f_I} N_{f_{II}} \tag{28}$$

where

N_0 = number of moles of oxidant

N_f = number of moles of fuel

I,II = regions I and II

Allowing the total fuel and air to be divided between regions I and II requires that the total moles still be conserved.

$$N_{0_I} = Z_0 N_{0_t} \tag{29}$$

and

$$N_{0_{II}} = (1 - Z_0) N_{0_t} \tag{30}$$

with

$$N_{f_I} = Z_f N_{f_t} \tag{31}$$

$$N_{f_{II}} = (1 - Z_f) N_{f_t} \tag{32}$$

where

Z_0 = mole fraction of total oxidant $0 < Z_0 < 1$

Z_f = mole fraction of total fuel $0 < Z_f < 1$

$Z_0 \neq Z_f$

The total air-fuel ratio of the charge in the displacement volume is given by the expression

$$AF_t = \frac{(N_{0_t})(MW_0)}{(N_{f_t})(MW_f)} \tag{33}$$

where

AF_t = total air fuel ratio, lbm air/lbm fuel

N = total number of moles

MW = molecular weight, lbm/lbmole

Likewise the air-fuel ratios in regions I and II are equal to

$$AF_I = \frac{(N_{0_I})(MW_0)}{(N_{f_I})(MW_f)} \tag{34}$$

and

$$AF_{II} = \frac{(N_{0_{II}})(MW_0)}{(N_{f_{II}})(MW_f)} \tag{35}$$

Combining Equations (33) and (34) into Equation (31) and likewise Equations (33), (35) and (32) gives

$$AF_I = \frac{Z_0}{Z_f}(AF_t) \tag{36}$$

and

$$AF_{II} = \frac{(1 - Z_0)}{(1 - Z_f)}(AF_t) \tag{37}$$

The mass balancing ratio β_M can be expressed in terms of the molar fuel-air analysis as

$$\beta_M = \frac{M_I}{M_{II}} = \frac{(N_{0_I} + N_{f_I})(MW_I)}{(N_{0_{II}} + N_{f_{II}})(MW_{II})} \tag{38}$$

The corresponding molecular weights of regions I and II are equal to

$$MW_I = \left[\frac{N_{0_I}}{N_{0_I} + N_{f_I}}\right] MW_0 + \left[\frac{N_{f_I}}{N_{0_I} + N_{f_I}}\right] MW_f \tag{39}$$

and

$$MW_{II} = \left[\frac{N_{0_{II}}}{N_{0_{II}} + N_{f_{II}}}\right] MW_0 + \left[\frac{N_{0f_{II}}}{N_{0_{II}} + N_{f_{II}}}\right] MW_f \tag{40}$$

Substituting Equations (39) and (40) into (38) yields an expression for β_M in terms of the moles of fuel and air in regions I and II as

$$\beta_M = \frac{N_{0_I} MW_0 + N_{f_I} MW_f}{N_{0_{II}} MW_0 + N_{f_{II}} MW_f} \tag{41}$$

Using the definition of AF_I and AF_{II} found in Equations (34) and (35)

$$\beta_M = \frac{N_{0_I} MW_0 + (N_{0_I} MW_0/AF_I)}{N_{0_{II}} MW_0 + (N_{0_{II}} MW_0/AF_{II})} \tag{42}$$

$$\beta_M = \frac{N_{0_I}}{N_{0_{II}}}\left[\frac{1 + (1/AF_I)}{1 + (1/AF_{II})}\right] \tag{43}$$

Since

$$N_{0_I} = Z_0 N_{0_t} \tag{29}$$

and

$$N_{0_{II}} = (1 - Z_0)N_{0_t} \tag{30}$$

with

$$AF_I = \frac{Z_0}{Z_f}(AF_t) \tag{36}$$

and

$$AF_{II} = \frac{(1 - Z_0)}{(1 - Z_f)} AF_t \tag{37}$$

then

$$\beta_M = \left[\frac{Z_0}{1 - Z_0}\right]\left[\frac{1 + (Z_f/Z_0)(1/AF_t)}{1 + (1 - Z_f/1 - Z_0)(1/AF_t)}\right] \tag{44}$$

Equation (44) expresses the mass balancing ratio β_M in terms of the overall air-fuel ratio and the mole fractions of fuel and oxidant in region I. An expression for the heat balancing ratio β_H can also be written in terms of the above parameters as

$$\beta_H = \frac{Q_{II}}{Q_I} = \frac{(N_{f_{II}})(MW_f)(HV)}{(N_{f_I})(MW_f)(HV)} \tag{45}$$

$$\beta_H = \frac{N_{f_{II}}}{N_{f_I}} = \frac{(1 - Z_f)(N_{f_t})}{Z_f N_{f_t}} = \frac{1 - Z_f}{Z_f} \tag{46}$$

Numerical computations using the molar analysis for a stoichiometric C_8H_{18}-air mixture were obtained using Equations (36), (37), (44) and (46). The results are found in Table VII and shown graphically in Figures 7 and 8. Table VI and Figure 7 show that the air fuel ratio for region I, i.e., composition associated with constant total volume heat addition, is rich for all values of β_M and β_H less than one and lean for all values of β_M and β_H greater than one. Table VI and Figure 8 further indicate that the air-fuel ratio for region II, i.e., mass associated with constant pressure heat addition, is lean for all values of β_M and β_H less than one and rich for all values of β_M and β_H greater than one.

The current results are all related to ideal Otto or spark ignition engines operating at an overall stoichiometric air-fuel ratio. Since real engine performance for the heat balanced engine[2] has shown best power at an air-fuel ratio of about 16 and best thermal efficiency at an air-fuel ratio of about 20, it would be extremely useful to extend the current fuel-air calculations to these overall composition limits. Moreover, since the cycle is applicable to compression ignition as well as spark ignition, the fuel-air approach should also be extended to compression ignition.

TABLE VII. QEHBC AIR FUEL RATIOS
vs. BALANCING RATIOS - MOLAR ANALYSIS

Z_0	Z_f	AF_I	AF_{II}	β_H
		β_M = 0.6 →		
0.35	0.7530	7.028	39.789	0.3280
0.36	0.6018	9.045	24.301	0.6617
0.375	0.375	15.12	15.12	1.6667
0.38	0.2994	19.190	13.381	2.3400
0.39	0.1482	39.789	10.828	5.7476
		β_M = 0.8 →		
0.42	0.8140	7.802	47.148	0.2285
0.43	0.6628	9.809	25.559	0.5088
0.44	0.5116	13.004	17.337	0.9547
0.4444	0.4444	15.12	15.12	1.2500
0.45	0.3604	18.879	13.002	1.7747
0.46	0.2092	33.247	8.992	3.780
		β_M = 1.0 →		
0.50	0.50	15.12	15.12	1.000
		β_M = 1.2 →		
0.53	0.7791	10.2854	32.1741	0.2835
0.54	0.6279	13.0028	18.6931	0.5925
0.5454	0.5454	15.12	15.12	0.8333
0.55	0.4767	17.4439	13.0028	1.0976
0.56	0.3255	26.0107	9.8637	2.0719
0.57	0.1743	49.4373	7.8743	4.7362
		β_M = 1.4 →		
0.57	0.7849	10.9798	30.2306	0.2740
0.58	0.6337	13.8380	17.3382	0.5780
0.5833	0.5833	15.12	15.12	0.7143
0.59	0.4825	18.4874	12.272	1.0724
0.60	0.3313	27.3803	9.0449	2.0181
0.61	0.1801	51.2022	7.1924	4.5515

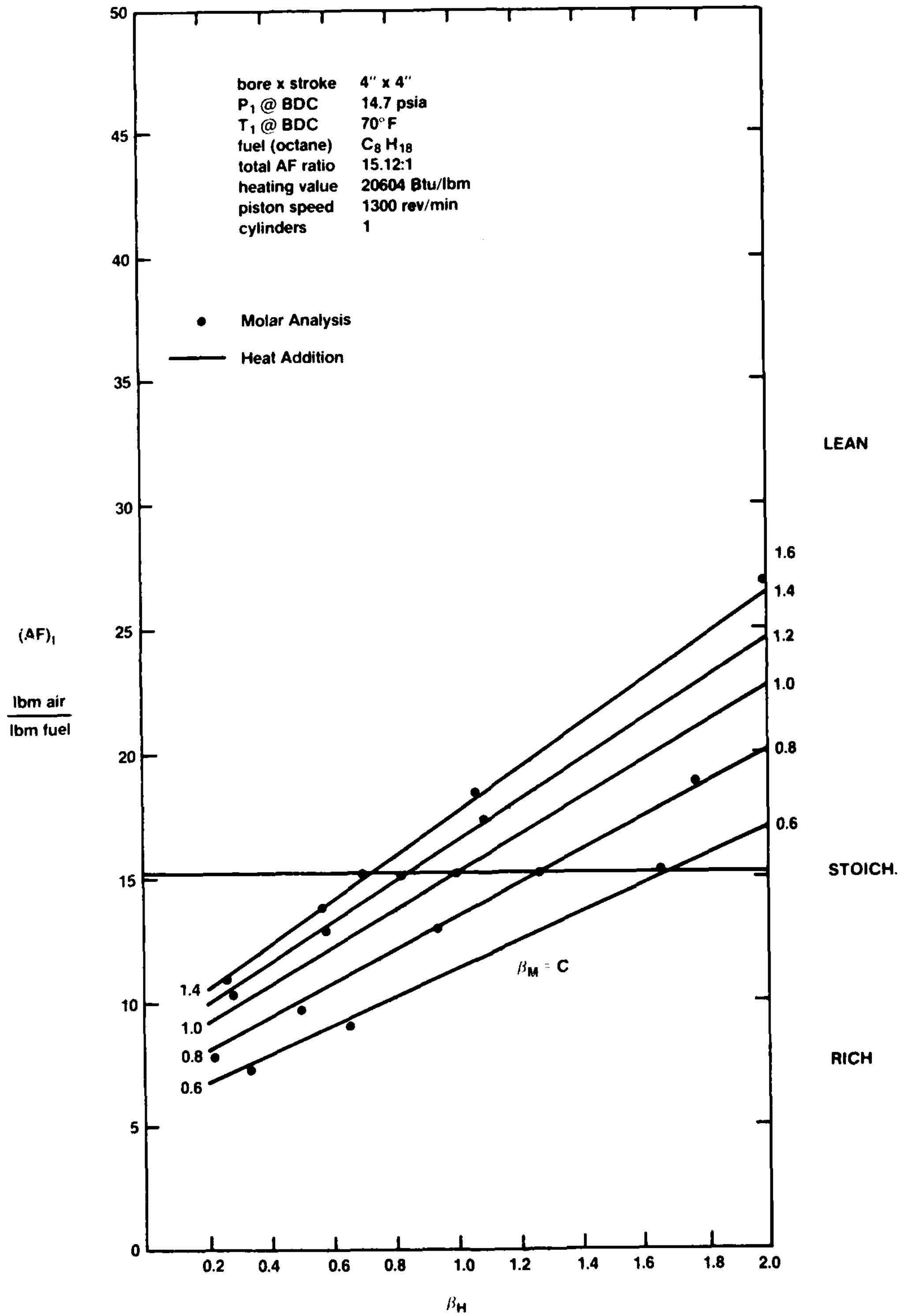

Figure 7 AF — REGION I VS BALANCING RATIOS (MOLAR ANALYSIS)

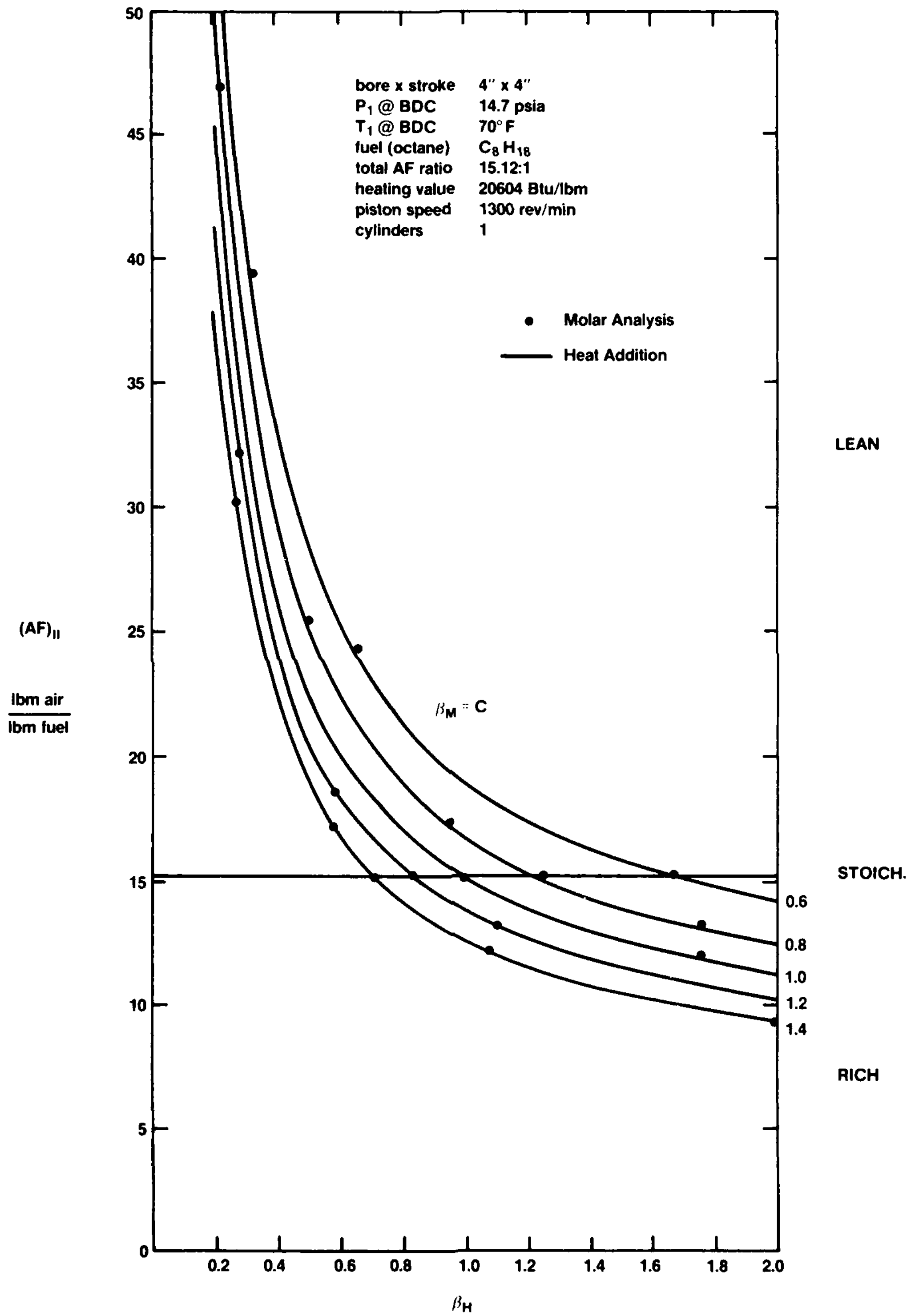

Figure 8 AF — REGION II VS BALANCING RATIOS (MOLAR ANALYSIS)

V. CONCLUSION AND RECOMMENDATIONS

The quasi-equilibrium air standard heat balanced cycle, QEHBC, has been modified to investigate the influence of the fuel-air thermochemistry on predicted cycle behavior. Based on this study several important conclusions can be stated. All results compare Otto and QEHBC cycles having equal engine geometries, total heat addition, compression ratios, overall stoichiometric air fuel ratios, and engine speed.

1. For stoichiometric heat addition the indicated output of the heat balanced cycle can exceed that of the corresponding Otto cycle.
2. For stoichiometric heat addition the indicated thermal efficiency of the heat balanced cycle can exceed that of the corresponding Otto cycle.
3. For stoichiometric heat addition the indicated specific fuel consumption of the heat balanced cycle is less than that of the corresponding Otto cycle.
4. These results occur for the optimum values of the mass balancing ratio, β_M, and heat balancing ratio, β_H, previously developed.[8]
5. The thermodynamic heat balancing ratios, β_M and β_H, are not independent parameters but are both functions of the overall total air fuel ratio and the corresponding air fuel ratios in the two mass balancing regions I and II.
6. For overall stoichiometric proportions the QEHBC air fuel ratios in regions I and II can be varied over a wide range of values and in general are not stoichiometric.

7. The indicated performance of the heat balanced cycle can be dramatically changed by simply varying the air fuel ratios in the two balancing regions I and II.

8. Optimum values for the heat balancing ratios of β_M and β_H suggest that the air fuel ratio associated with constant total volume heat addition be rich.

9. Optimum values for the heat balancing ratios of β_M and β_H suggest that the air fuel ratio associated with constant pressure heat addition be lean.

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